DESIGN CRITERIA FOR CO_2 EVAPORATORS

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Introduction

Following the replacement of CFCs and HCFCs by chlorine-free refrigerants because of their damaging effect on the ozone layer, concerns began to be raised at an early stage about the high risks posed by the new HFC refrigerants and their effect on exacerbating the greenhouse effect. In recent years, attention has therefore increasingly been focused on natural refrigerants. The past few years have witnessed a significant increase in the use of carbon dioxide (CO_2) and, particularly since the 2001 conference hosted by the German Association of Refrigeration and Air Conditioning Technology (Deutsche Kälte- und Klimatechnik Verein, DKV) in Ulm, if not earlier, it was recognised that CO_2 has applications that are now economically viable—principally in ammonia (NH_3) cascade operations. Plants of this type generally operate at evaporating temperatures of between –40 and –50 °C. There are some special features which must be borne in mind in designing CO_2 evaporators, and these will be examined in this article.

Classification of CO_2 evaporators

Essentially the main difference between different types of CO_2 evaporators lies in their mode of operation, i.e., direct expansion evaporators, pump-operated evaporators and evaporators for generating process gas. From a thermodynamic viewpoint, they may be classified between the two variants: pump operation and direct expansion (Figure 1). A further distinguishing feature is the required operating pressure. Provided it has been confirmed at the planning stage that the operating pressure of 32 bar (the current nominal pressure for Güntner evaporators) will not be exceeded, any standard materials can, in principle, be used. This is of particular importance in cases where the direct expansion system is being employed because internally grooved tubes can be used in such cases. This variant is the most economical in all cases.

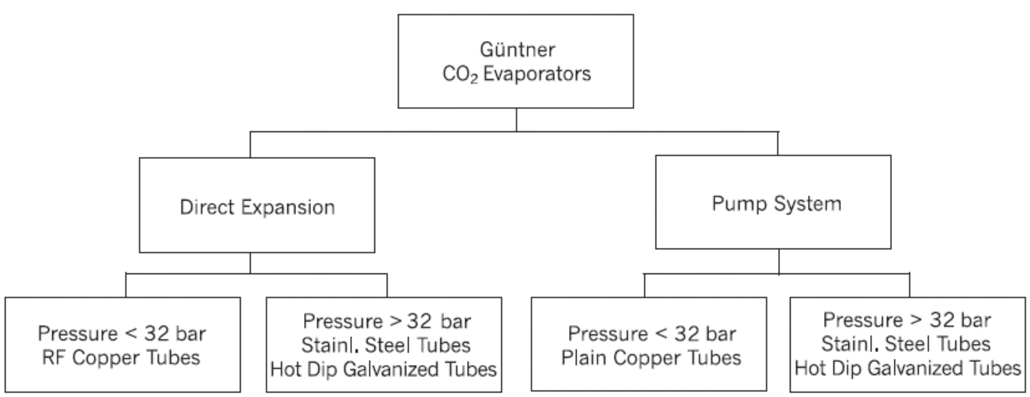

Figure 1: Overview of CO_2 evaporator types

Where hot gas defrosting using CO_2 is being carried out, the permitted pressure is generally between 45 to 50 bar. This is difficult to achieve with copper tubes; thicker heat exchanger tubes would have to be used, and these are not available with internal grooves. In addition, the joining tubes (header outlets, connecting tubes) would also require thicker walls and would need to be purpose-built. Heat exchangers made of hot dip galvanised or stainless steel—where pressures of 50 bar are relatively easy to achieve—would be suitable in these cases.

Design of CO_2 evaporators

The CO_2 evaporator is designed by using thermodynamic theory and engineering principles, ranging from mass flow rate, heat transfer, and refrigerating capacity, to selection of materials for tubing and design of fin structures.

However, when CO_2 is being used, an additional factor comes into play: the internal heat transfer with CO_2 is so high that its effect on the evaporating capacity is practically negligible. For example, an increase in heat transfer at the refrigerant side from 2,000 W/m^2 K to 3,000 W/m^2 K in a typical industrial air cooler increases the overall heat transfer coefficient by only about 6%.The effect on refrigerating capacity is even less because a greater heat transfer value leads to increased air cooling and, thus, to a lower driving temperature difference. Some of the internal heat transfer coefficients occurring when CO_2 is being used are still considerably greater than 3,000 W/m^2 K, so any uncertainty about the calculations in this respect may be ignored.

Special properties of CO_2 in comparison with R22 and NH_3

As is widely known, the special property of CO_2 as a refrigerant is its high saturation pressure. On the one hand, this property may lead to considerable reluctance to use CO_2 as a refrigerant because it makes it rather difficult to handle, especially at higher temperatures. However, at low temperatures, this property is a decisive advantage. For example, a comparison of the properties of CO_2 with those of R22 and NH_3 at a temperature of -40°C shows that the volumetric refrigerating capacity of CO_2 is about 7 times greater than that of R22, or 8 times greater than that of NH_3 (see Table 1). In other words, the tube cross-sections required in a CO_2 plant are only about one-eighth of those in a comparable NH_3 plant. This leads to smaller tube cross-sections and smaller compressors, which in turn lead to lower refrigerant charge sizes and, ultimately, to more economical plants. However, the high pressure and the associated high gas density also lead to the above-mentioned high temperature heat transfer during evaporation.

Table 1: Material properties of CO_2, R22 and NH_3 at - 40°C

Refrigerant	CO_2	R22	NH_3
Vapour pressure [bar]	10	1	0.7
Enthalpy of evap. [kJ/kg]	322	243	1387
Density of gas [kg/m^3]	26.24	4.85	0.64
Gas volume flow for 10 kW [m^3/h]	6	41	47
dp/dT [bar/K]	0.37	0.05	0.04
Required distributions (ca. 8 m/s)	2	12	12

A further consequence of the high pressure level is the weak temperature/pressure dependency. For example, a pressure change of about 0.37 bar is required to change the vapour pressure by 1 K at -40 °C. When using R22, this could be achieved with a pressure change of as little as 0.05 bar, or with just 0.04 bar using NH_3. Because of this beneficial side effect, pressure losses have only a negligible effect on the evaporating temperature. However, this property is ultimately the only thing that makes it possible to use CO_2 with normal evaporator geometries. In order to explain this more clearly it is necessary to illustrate the relationships by means of a specific example: an evaporator with 6 rows of tubes in the air flow direction, 12 rows of vertical tubes and a tube diameter of 15 mm would give the following theoretically possible circuiting variants (pass numbers); assuming a fixed refrigerating capacity of 15 kW, a refrigerant speed of over 7 m/s can be achieved with R22 at -40°C with a 6-pass circuiting (12 distributions). However, it is only possible to achieve a speed barely exceeding 6 m/s with a 36-pass circuiting (2 distributions) when using CO_2. This means that, when CO_2 is being used, the distribution length must be 6 times longer. Assuming on an initial estimate that the pressure loss per metre of tube is the same for all refrigerants at the same speed, there would be 6 times more pressure loss with CO_2. In fact the actual pressure loss using CO_2 is even slightly greater because of the properties of the materials (see Figure 2).

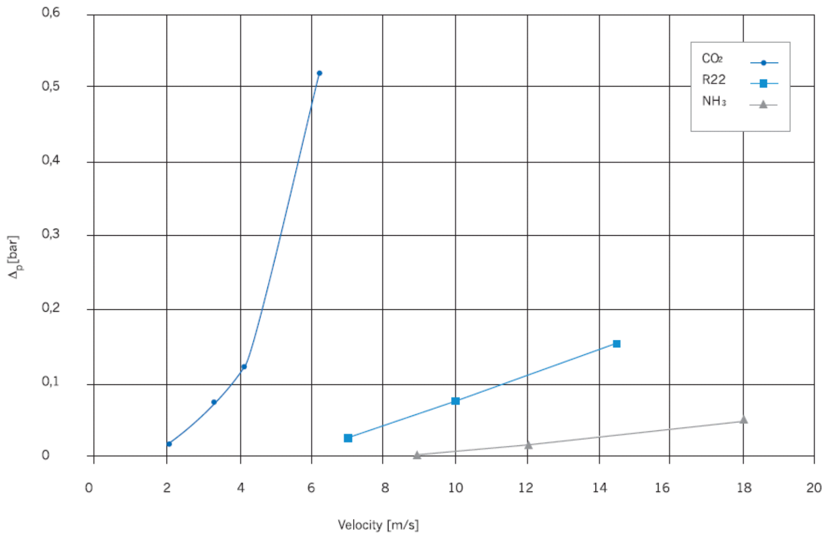

Figure 2: Pressure loss in relation to refrigerant velocity

However, from a thermodynamic viewpoint, it is only the temperature drop caused by the pressure loss that is important, since this lowers the driving temperature difference at the evaporator and, as a result, the refrigerating capacity. This is however only slightly greater than with R22 and NH_3 because of the weak temperature pressure/dependency (see Figure 3). If this were not the case, other heat exchanger geometries would have to be used for CO_2, i.e., tubes with a considerably smaller diameter would have to be used without increasing the number of tubes.

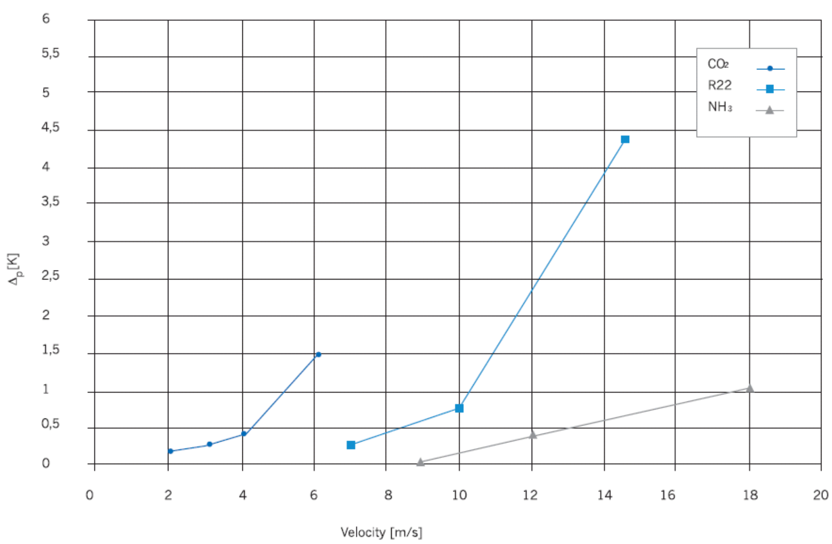

Figure 3:"Temperature loss" in relation to refrigerant velocity

Results of design calculations

By way of an example, let us assume that a comparison of the three refrigerants, CO_2, R22 and NH_3, was to be carried out using the evaporator described above. The conditions chosen would be:

t_o = -40°C (evaporating end temperature)
t_1 = –30°C (air intake)
RH = 95% (relative humidity of air)

In order to exclude other influences, the same quantity of air was used in all cases. Similarly, the calculations were based on the use of copper tube throughout, even though this was purely a theoretical assumption for NH_3. The capacity was set at a constant value. Circuitry appropriate to the respective refrigerant was used.

Figure 4 shows the internal heat transfer coefficients in relation to the refrigerant speed using direct expansion. As can be seen, values can be achieved with CO_2 with plain tube that can only be achieved with internally grooved tube when using R22. In this comparison, it should be noted that the values for NH_3 are purely theoretical since, as previously mentioned, the calculations were based on the use of copper tube and on superheating of 5 K, which is difficult to achieve with NH_3. However, the fact that the temperature drop caused by pressure loss is more important than the refrigerant speed makes Figure 5 more meaningful, showing as it does the internal heat transfer coefficient in relation to the "temperature drop" caused by pressure loss.

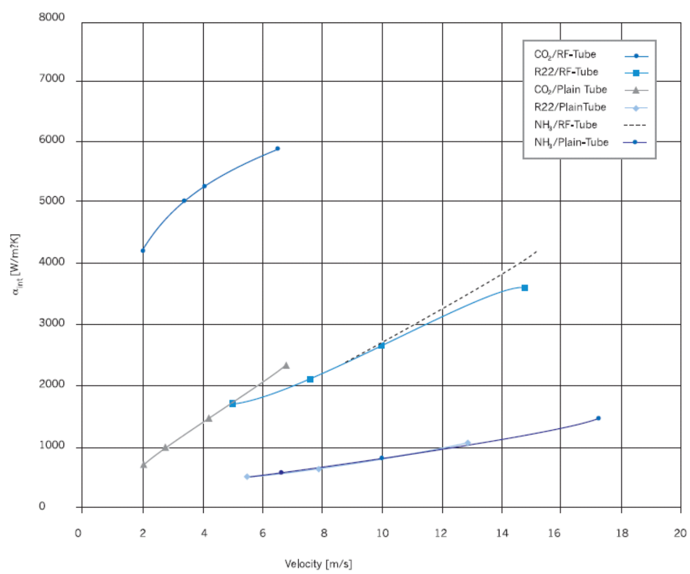

Figure 4: Heat transfer of different refrigerants in relation to velocity

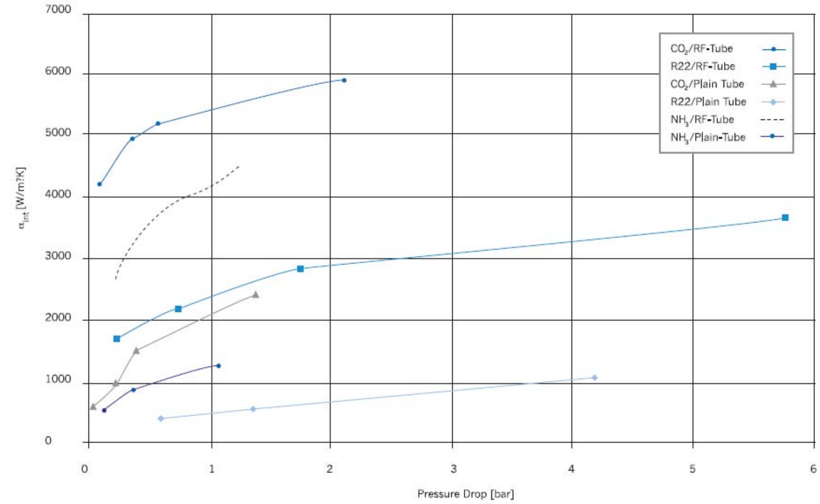

Figure 5: Heat exchange of different refrigerants in relation to pressure loss

However, Figure 5 does not provide much help in creating an optimum design for an evaporator either. What ultimately matters is the product of the total heat transfer coefficient (U) and the average logarithmic temperature difference (ΔT), which is often also expressed as the thermal load. Figure 6 shows this again for the three refrigerants (CO_2, R22 and NH_3) for evaporators with plain tubes and internally grooved tubes. This clearly shows that the optimum speed for CO_2 is markedly less than the speed achievable with R22, and substantially less than the speed achievable with NH_3. This is due to the greater pressure loss of CO_2 (because of the circuitry) and to the greater heat transfer. Both result in a situation where, when using CO_2 even at relatively low speeds and on increasing the speed, the negative effect of the pressure loss has a greater effect on capacity than the positive effect caused by the increasing heat transfer. Exactly the opposite situation applies when using NH_3, where good heat transfers can be achieved only at very high speeds. However, to make up for this, the pressure losses at these speeds remain relatively low because of the circuitry. However, it must be stressed again that the values for NH_3 shown in Figure 6 are practically unachievable. With this in mind, the advantages of using CO_2 in pump-operated evaporators are even more evident.

Figure 7 shows a comparison of the internal heat transfer coefficient at a pumping rate of 2.5. The internally grooved tube has been dispensed with here, since it produces no significant advantages when pump operation is being used. In this case, calculations for NH_3 were also based on an assumption that copper tube was used. The difference between the other refrigerants and CO_2 would therefore be even greater in a real heat exchanger.

Figure 8 again shows the thermal load of the different variants. The values for direct expansion are shown again for comparison purposes. It can be seen that CO_2 excels in pump operation. Again, this is because when CO_2 is being used, the driving temperature difference plays a more important role than the heat transfer coefficient (which is high in any event). Because of the absence of superheating, the average logarithmic temperature difference using pump operation is generally greater than when using direct expansion. The total thermal load during pump operation is therefore greater than when using direct expansion, although there is somewhat less heat transfer. When using NH_3, the same effects again result in the opposite outcome: the greater heat transfer with direct expansion outweighs the disadvantage caused by the smaller temperature difference such that direct expansion has an advantage, at least in this theoretical case.

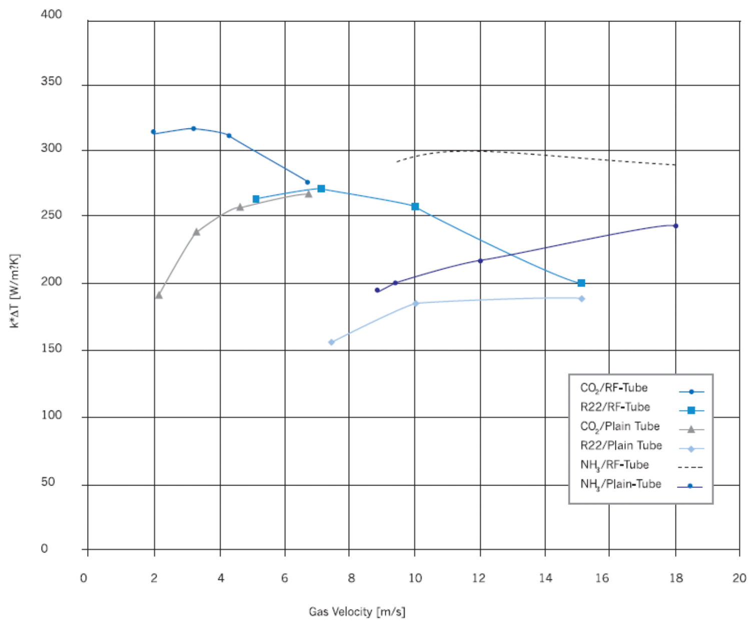

Figure 6:Thermal load in relation to refrigerant speed

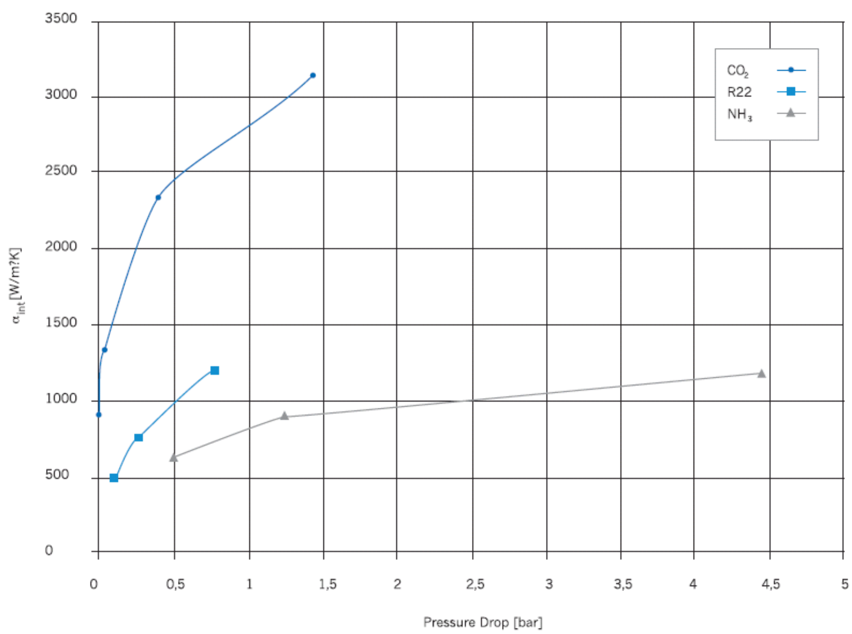

Figure 7: Internal heat transfer with pump operation

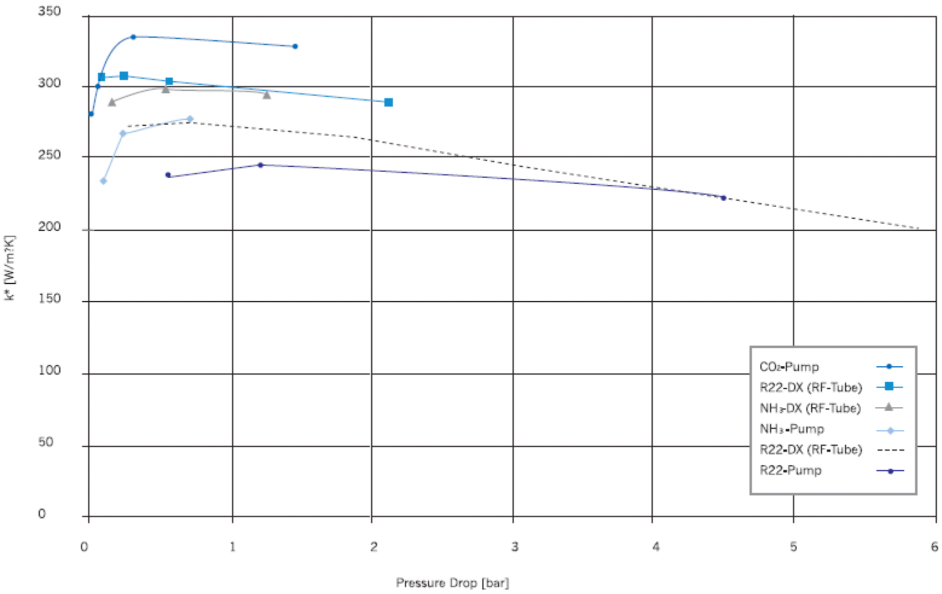

Figure 8: Heat load of various refrigerants in different modes of operation

Summary

By using CO_2, greater heat transfer coefficients can be achieved both in pump operation and in direct expansion than with all other refrigerants currently available. Because of the high pressure level and the resulting weak pressure/temperature dependency, all standard heat exchangers can be used just by altering the circuiting to an increased number of passes or smaller distributions. Should operating pressures of 32 bar be insufficient, either thicker walls or other materials (i.e., steel, stainless steel) must be used. In that case, the cost advantage achieved by the greater heat transfers is again offset or even reversed. It would therefore be sensible in the long term to introduce a specific fin geometry which has a smaller tube diameter with the same tube spacing. In this way, the long distribution lengths could be dispensed with. Moreover, there would be the advantage that tubes of smaller diameters have greater stability under pressure and that the required wall thicknesses would thus be kept within reasonable bounds. However, it is only worth investing in such a tool if greater quantities of CO_2 evaporators are required.